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POWER LOSSES IN THE TRANSMISSION MACHINERY OF CENTRAL STATIONS.

BY WM. S. ALDRICH, MORGANTOWN, W. VA.

(Member of the Society.)

THIS paper had its unexpected origin in correspondence with the Secretary of the Society after Volume XIV. of the *Transactions* had been completed. It was deemed advisable to add some memoranda on the subject of power losses in the transmission machinery of central stations, which had come within the author's experience, and some of which were incompletely referred to in the discussion of Paper No. DXLVI., on the "Performance of Street Railway Power Plants," presented at Chicago meeting (July, 1893), by Messrs. Wm. A. Pike and T. W. Hugo.

The contents of the present paper are outlined below:

(1) A revision of the author's discussion of Paper No. DXLVI.

(2) An analysis of Paper No. DXLVI., with reference to the power losses in the transmission machinery.

(3) A memorandum of the power losses in a turbine plant for an electric railway.

(4) A memorandum of the power losses in a cable plant.

(5) Results of some tests to determine the power losses, in transmitting through a countershaft.

(1) REVISED DISCUSSION OF PAPER NO. DXLVI.

Table XVI., page 1146, Vol. XIV., of the *Transactions* of this Society, of the combined horse-powers of the St. Paul Test, No. 2, from cards taken every ten minutes, cannot possibly show the real variations of power in an electric railway plant. The frictional losses in the transmission machinery are best determined by special tests, before or after the regular run, having this one object in view. In this way the losses in the engine, transmission machinery, shafting and empty cables were determined in a joint test of a cable railway plant, made by Mr. Hermann S. Hering and the author.

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* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

It is that which leads me to consider the statement made on pages 1099–1100, regarding the loss in friction as obtained by taking the difference between the mechanical and electrical horse-power. It is true that such is the usual method pursued. In a joint test of an electric railroad, made in 1891, by Mr. Hermann S. Hering and the author, they found that it was impossible to arrive at any satisfactory conclusion as to what became of the motive power generated to drive the dynamos, when throwing the load on and off again as quickly as it is done in an electric railroad. At times there would be as much as 40 H. P. unaccounted for; that is, for a few seconds the dynamo output would fall more than 80 H. P. below the then delivered turbine horse-power; while, at other times, the horse-power delivered by the turbines and the dynamos would be equal. Under light load conditions, which remained steady for about two minutes at one portion of the run, the actual difference between the turbine horse-power and dynamo horse-power, delivered, was 43.8. All at once, while the turbines were decreasing their output, there would be 50 or 60 H. P. thrown off or on the dynamos, and the turbine gates would keep on closing. At other times, the same thing would happen while the turbines were increasing their output, and the turbine gates would keep on opening, causing the difference of power between that delivered by the turbines and that delivered by the dynamos to the line to run enormously high; the maximum difference so observed was 140 H. P., or almost 100 H. P. in excess of that lost in the transmission machinery and dynamo.

Between these extremes one can readily see that the transmission machinery acts in the way of a fly-wheel, storing up the excess of energy on the one hand, and giving out part of its stored energy during a deficiency on the other hand. These great differences of power exist only for a few seconds. The dynamo load does not go down or up to a definite point and stay there, so permitting the turbine to settle to its new load; but, while the turbines are seeking to adjust themselves to the new requirements, the dynamo load has either gone farther in the same direction, or immediately changed about to the opposite condition. We took the ground that these gross irregularities were largely inherent in a turbine plant.

In writing our report on this test for the *Electrical World* (May 28 to July 30, 1892), we stated that steam-engines would not be at all likely to show the same results of great excesses and deficiencies of output, while the dynamo load continued to vary in its characteristic fashion for street railway purposes. We are greatly surprised, therefore, to find that the tests of Messrs. Pike and Hugo, on the Corliss engines of the Minneapolis and St. Paul electric railway plants, do show up much the same as in our turbine tests referred to. They even note that, when lightly loaded, the difference between the engine and dynamo output is *minus*, showing that the dynamos are delivering more power than the engine, similar to the case we have cited for the turbine plant. While they believe this peculiarity to be due to imperfect (better, *tardy* regulation) we think it is due almost entirely to the fly-wheel action of the transmission machinery. We took the ground that a steam-engine ought to be able to regulate itself more quickly and closely than a turbine, on account of the elastic nature of the working fluid. Probably it would, if it did not have such a drag upon it as heavy transmission machinery and countershafting, in addition to being driven by an incompressible working fluid.

Steam-engines are supposed to be capable of closer and more accurate regulation than turbines, by reason of this very different nature of the working fluid employed. The regulation test of the high-speed automatic engines, noted on page 1116, shows that from a very heavy load to a mere friction load the engine regulated in four revolutions, or one second of time. It is believed that still quicker regulation will come to be the ordinary practice in high-speed automatic engines.

Now, I think it is a very important point to bear in mind that a high-speed automatic engine may be made to meet the rapidly changing loads in an electrical power plant with an almost equal economy. If we take the two papers presented at this meeting—the Laketon test of a triple-expansion pumping engine, in which the load is quite uniform; and the present test of electric railway engines, in which the loads are very irregular—we find much food for thought.

The main point which I wished to bring out in this discussion

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is that the high-speed automatic engine, directly connected (or better, directly coupled), is best adapted to meet the sudden and very great variations in electric power plants, while the well-known type of Corliss engine seems best adapted to steady loads.

(2) ANALYSIS OF PAPER NO. DXLVI, WITH REFERENCE TO THE POWER LOSSES IN THE TRANSMISSION MACHINERY.

Table I., following, contains a brief presentation of the principal figures in this connection, from the paper under discussion. It is quite important to notice the increase of frictional losses in the St. Paul Test, No. 2, in which the power is transmitted to the dynamos, through the friction clutches on the counter-shaft. It is probable that there is great frictional loss in the use of certain forms of friction clutches, or in old and ineffective parts. In this connection, also, it is to be especially noted that this No. 2 test was one in which there were belt tighteners used. No doubt a great loss may be accounted for here, also. Both of these causes may have operated to raise the per cent. friction from 12.10 of No. 1, to 17.40 of the No. 2, as tabulated.

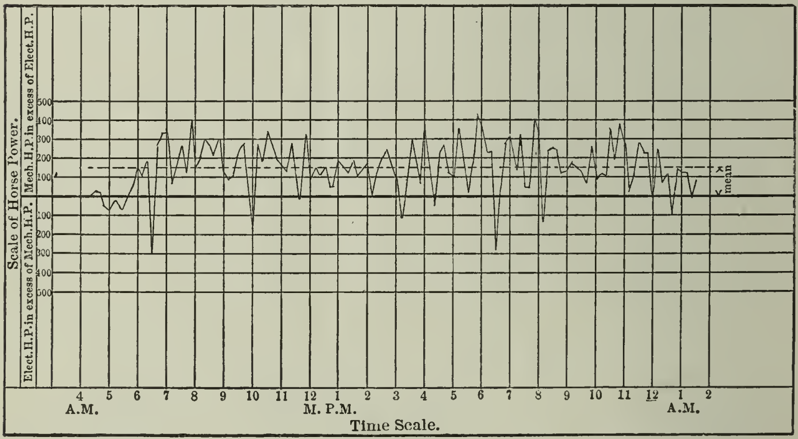

Fig. 204.—Plot of the Differences between Mechanical and Electrical Horse-Powers, for the Minneapolis Test, No. 1.

Fig. 204 contains a plot of the differences between the mechanical and electrical horse powers, of Fig. 387, opposite page 1098, as given by Messrs. Pike and Hugo for the Minneapolis

Test, No. 1. The characteristic variations are to be especially noted; sometimes the mechanical is in excess, sometimes the electrical. The dotted horizontal line shows the mean value of this difference—the value given on page 1097, for the mean loss by friction, of 149.9 H. P., and as given in Table I, for this test.

TABLE I.

ANALYSIS OF POWER LOSSES IN MINNEAPOLIS AND ST. PAUL TESTS, OF PAPER NO. DXLVI.

No. of Test.	Location of Plant.	Kind of Engine.	How connected to Dynamo.	Ind. H. P. delivered by engine.	Elec. H. P. delivered by dynamos.	Eng. Friction H. P. by Friction Run.	Per cent. Eng. Friction.	Total Friction H. P. = I. H. P. — Elec. H. P.	Per cent. Total Friction H. P.
1	Power House, No. 1, Minneapolis.	Tri-cylinder Cond. Corliss.	By Countershaft	1,237.90	1,088.0			149.90	12.10
2	Hill St. Power House, St. Paul.	Tri-cylinder Cond. Corliss.	By Countershaft carrying Friction Clutches.	756.20	624.7			131.50	17.40
2*d*	Same.	Same.	Same.	859.73				164.43*	19.12
3	31st St. Power House, Minneapolis.	Westinghouse Compounds.	Belted direct to Dynamo.	522.50	434.2	14.27	3.7†	88.30	16.90

(3) MEMORANDUM OF THE POWER LOSSES IN A TURBINE PLANT FOR AN ELECTRIC RAILWAY.

The following particulars of the performance of the turbine power plant of the Neversink Mt. Electric Railway, Reading, Pa., have been taken from the authors' reprint of the published report of the test, as it appeared in the *Electrical World*, May 28 to July 30, 1892. At the special request of this journal, Mr. Hermann S. Hering and the author made a series of tests of this plant, from August 17 to 22, 1891.

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* Friction horse-power of engine, transmission machinery, and mechanical and electrical losses in dynamos, obtained by indicating engine with no load on the dynamos.

† Per cent. engine friction, as compared with the maximum horse-power of any one engine.

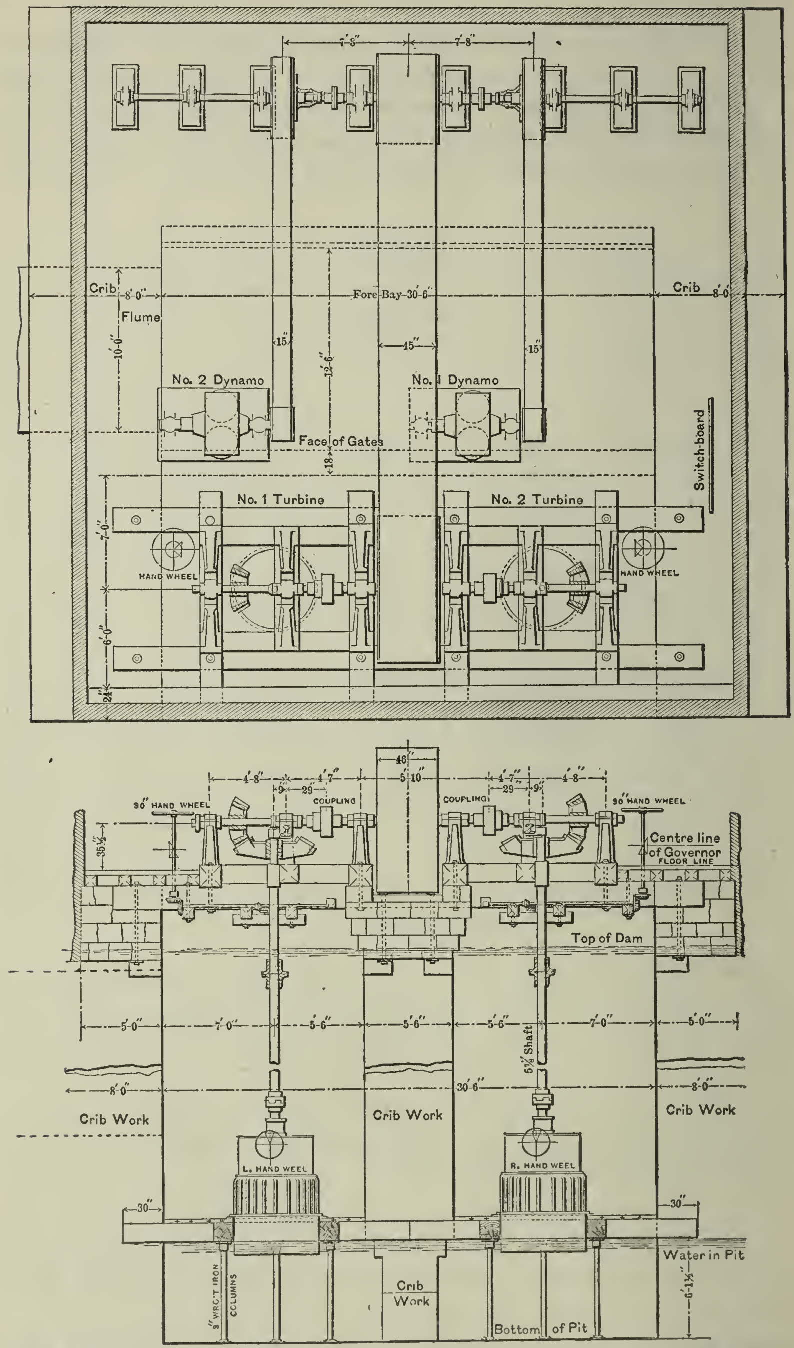


FIG. 205.—Plan and Elevation of the Turbine Plant of the Neversink Mountain Electric Railway, Reading, Pa.

The plan and elevation of the turbine plant are shown in Fig. 205.

Each vertical shaft of the two Hercules turbines transmits its power through a mortise crown and jack gear to the horizontal shaft carrying the main driving pulley. The power is thence transmitted to a counter shaft, from which each dynamo is belted by means of a friction clutch pulley.

Table II. contains the details of the turbine installation, and Table III. of the transmission machinery.

The turbine data for the test was obtained from the Holyoke Machine Company, through whose kindness the authors were furnished with a complete test-sheet of the actual performance of a 45-inch Hercules turbine, of exactly similar design to those here used.

Performance of the Turbine Plant under a Light Load.

In this, known in our report as the "Special Car Test," observations were made to determine the distribution of power, and the several losses from the hydraulic head of the water to the available power at the car axle, during the run of a single car over the road. The graphical log of the test, so far as the present paper requires, is shown in Fig. 206.

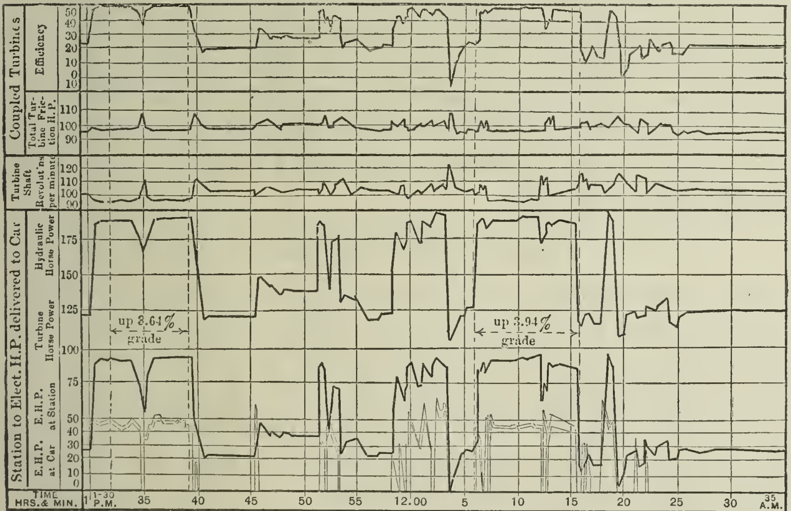
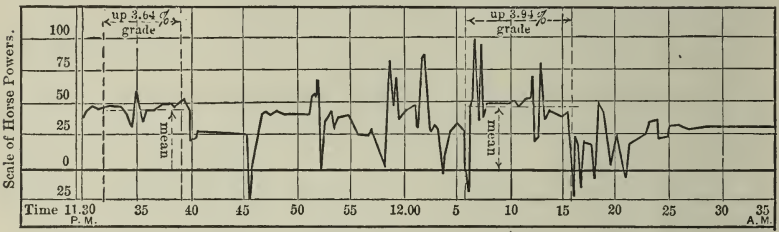


FIG. 206.—Graphical Log of the Performance of the Neversink Mountain Turbine Plant, under a Light Load.

Fig. 207 shows the variations of the losses from the turbine shaft to the dynamo terminals, obtained by taking the instantaneous differences between the turbine horse-power and dynamo output, and plotting on the same time base as Fig. 206. The horizontal dotted line shows the mean (from the integrated means) for the section marked off by the vertical dotted line.



CURVE SHOWING VARIATION OF LOSSES FROM TURBINE SHAFT TO DYNAMO TERMINALS.

FIG. 207.—Plot of the Differences between Mechanical and Electrical Horse-Powers for Reading Test; Light Load.

Table IV. contains the more interesting and important observed and mean values, taken from Fig. 206, and Table V. the summary of the power distribution and efficiencies.

A noteworthy feature of this test is, that the No. 2 turbine was run throughout with a fixed gate at 17.4% proportional part of the full opening of the speed gate. The regulating was entirely done by the No. 1 turbine. Anomalous conditions frequently arise when one of the turbines is thus operated at a fixed gate, as will be readily seen upon an examination of the graphical log, Fig. 206.

Performance of the Turbine Plant under a Heavy Load.

This is designated the "Test of the Power Plant" in our report, and observations were made similar to those referred to in the special car test. The graphical log, as far as relates to the present paper, is shown in Fig. 208, and the variations of the power losses from the turbine shaft to the dynamo terminals in Fig. 209, in which the horizontal dotted line shows the mean (from the integrated means), as before.

Table VI. contains the observed and mean values from Fig. 208, and Table VII. the summary of the power distribution and efficiencies. The time base of the graphical log was divided into two sections, each of which possessed typical groups of changes, and the several values are given for these sections, as well as for the entire run.

In each of these tests there were marked peculiarities in the variations of the power losses between the turbine and the dynamo output, as noted in the discussion. It was believed that these were inherent in turbine plants only, but from a similar plot of the Minneapolis Test, No. 1, given in section (2), it is apparent that the same difficulties are being experienced in electric railway plants driven by Corliss engines.

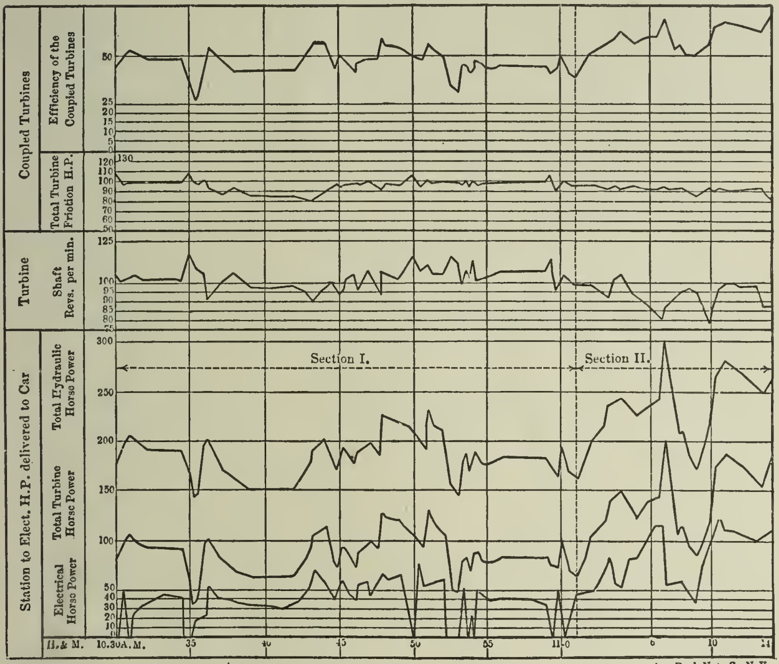


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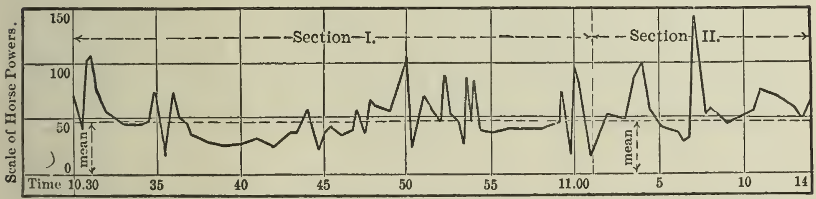
FIG. 208.—Graphical Log of the Performance of the Neversink Mountain Turbine Plant, under a Heavy Load.

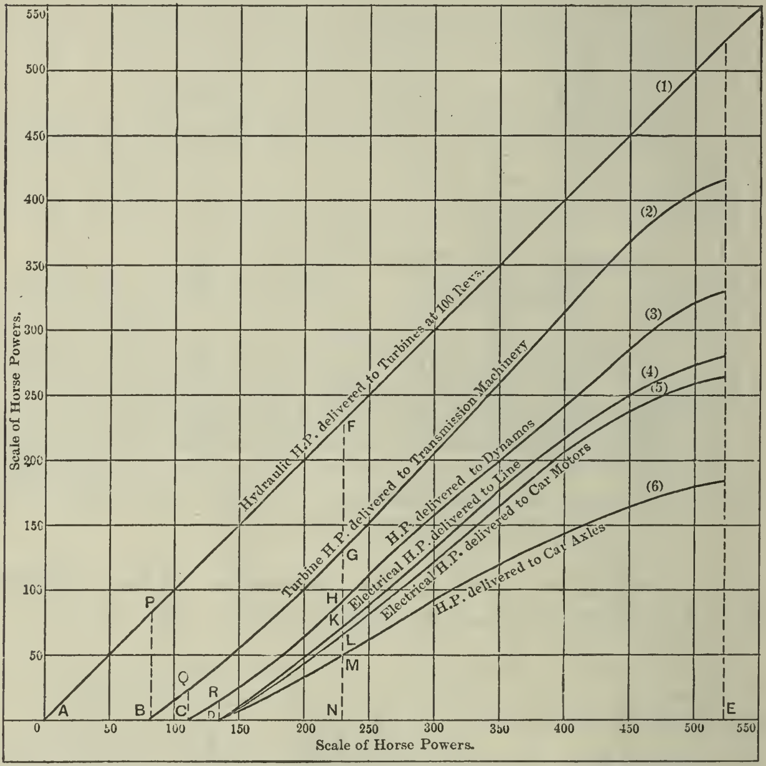


CURVE SHOWING VARIATION OF LOSSES FROM TURBINE SHAFT TO DYNAMO TERMINALS.

FIG. 209.—Plot of the Differences between Mechanical and Electrical Horse-Powers for Reading Test; Heavy Load.

Average Power Distribution of the Turbine Plant.

The mean values for the several losses in the distribution of the power (mean speed of turbines at 100 revolutions) have been plotted in the curves in Fig. 210. These do not represent the means, integrated or otherwise obtained from few or many observations, nor do they show just the state of affairs at any one moment. They are rather to be considered as general average values showing, by simple inspection, the proportional distribution of losses, and the final power available at the car axle for the various horse-powers up to the maximum hydraulic capacity of the station. It would, of course, be possible to obtain such a distribution of the power as is here shown, if the turbine speed could be maintained constant for such a length of time that the station performance would settle down to steady conditions, on



CURVES OF POWER DISTRIBUTION FROM HYDRAULIC H.P. TO CAR AXLE H.P.

FIG. 210.

the one hand; while the distribution and load of the cars out on the road should be almost uniformly maintained, on the other hand. The disturbing element in the station performance as here represented is the variable turbine speed; and with regard to the road conditions, it is the varying position of the cars *en route*, and the changing loads on the motors due to varying grades and the somewhat irregular passenger traffic. However, taking all these matters into consideration, the curves convey a very good idea of the relations between the successive losses.

The diagonal line marked (1) represents the H. H. P. delivered to the turbines at 100 revolutions, and for which either the horizontal or vertical scale of horse-powers may be used. The curve (2) is the T. H. P. delivered from the turbine shaft to the transmission machinery. It has been obtained for the several hydraulic horse-powers from the curves which the authors plotted from the data furnished by the Holyoke Machine Company. The horizontal distance, *A B*, Fig. 210, represents the H. H. P. required to drive the two turbines with no output; and that for any other turbine output, as *G N*, is measured by the horizontal distance, *A N*. It is to be carefully noted that the horizontal and vertical distances for any points, as *A B* and *B P*, or *A N* and *N F*, are equal only when referring to points on the diagonal line (1) of H. H. P.

The curve of horse-powers delivered to the dynamos (3) was obtained from a curve of power losses in the transmission machinery at the constant speed given above, and which losses have been set off from the points along (2) for any given horse-power delivered from the turbine shaft. The losses in the transmission machinery were obtained partly from observations made during the test and from calculations based on the dynamo output and efficiency. The point *C*, and its ordinate *Q C*, refer to conditions of the station test, when all the available T. H. P. was absorbed in the transmission machinery, the dynamos being unclutched.

The curve (4) of E. H. P. delivered to the line shows the dynamo output as measured at the switchboard, and its vertical distance, as *H K*, at any point, represents the total mechanical and electrical losses in the dynamos, for that proportional part of the dynamo output. As before, the point *D* and its ordinate *D R* correspond to conditions in which all the available T. H. P. is absorbed in the transmission machinery and dynamo losses, fields excited but zero amperes at the switchboard. The efficiency of the dynamos at any particular output is shown by the ratio of the respective ordinates of (4) and (3), as $N\ K$ divided by $N\ H$. The losses represented by the vertical distance at any point between (3) and (4), as $H\ K$, comprise the following: Mechanical friction of the journals in the bearing and of the air resistance, 1.34 H.P., and was assumed fairly constant under uniform speed at the different loads; hysteresis, Foucault currents and losses on the shunt fields, 4.22 H.P., also fairly constant under the same conditions; and the variable $C^2\ R$ losses in the armature and the series field coils calculated from resistances with temperature corrections.

The E. H. P. delivered to the car motors, curve (5), has been obtained by setting off vertical distances, as $K\ L$, from (4) which are uniformly 6% of the ordinates of (4) as $K\ N$; this per cent. of line losses being the mean already referred to.

The horse-power delivered to the car axle, curve (6), was drawn by deducting 30% from the measured ordinates of (5), the combined efficiency of the motor and gearing having been taken at 70%. Finally, the ordinates of curve (6), as $M\ N$, represent the horse-power available at the car axle for traction at any particular point within the limits of the preceding curves.

TABLE II.

DATA OF TURBINE INSTALLATION.

Water supply	
Flume, diameter in feet	10
Nominal head of water, in feet	18
Hercules turbines.	
Diameter, in inches	45
Effective area of guide opening, square inches	623
Rated speed, revolutions per minute	110
Commercial rated performance, full gate.	
Rated discharge of each turbine, cubic feet per minute	8,833
Rated efficiency, full gate and rated speed, in per cent	80
Rated horse-power, at this efficiency, each turbine	240
Actual performance at full gate.	
Actual discharge, each turbine, cubic feet per minute	8,588
Actual efficiency, 110 revolutions, in per cent	80.7
Maximum dynamometric horse-power, each turbine	210

Actual performance, about half gate.

Mean position of speed gate, being the proportional part of the full opening of the speed gate in per cent	53.3
Actual discharge at critical speed for this gate, cubic feet per minute	5,528
Efficiency, critical speed 100, in per cent	75.37
Efficiency, rated speed of 110	74.4
Actual dynamometric horse-power of each turbine at the critical speed of 100 revolutions per minute	132.5
Actual dynamometric horse-power of each turbine at the rated speed of 110 revolutions per minute	128.5

TABLE III.

DATA OF TRANSMISSION MACHINERY OF TURBINE PLANT.

Geared and belted connections.

Number of teeth on mortised crown gear, vertical shaft	49
Number of teeth on jack gear, horizontal shaft	31
Diameter of main driving pulley, on jackshaft, in inches	108
Diameter of main driven pulley, on countershaft	66
Diameter of dynamo-driving pulleys (friction clutch)	60
Diameter of pulleys on dynamos	26
Main driving belt, extra double, width in inches	45
Dynamo belts, double, in inches	15
Length of main driving belt, in feet	83

Rated speeds.

Turbine shaft (vertical), revolutions per minute	110
Jackshaft over turbines	175
Countershaft	286
Dynamos	650
Speed of main driving belt, feet per minute	4,950
Speed of dynamo belts	4,300

Distance between shaft centre lines.

Between the vertical shafts of the turbines, in feet	16.5
Between the jackshaft and countershaft	30
Between the countershaft and dynamos	20

TABLE IV.

PERFORMANCE OF TURBINES AND DYNAMOS UNDER LIGHT LOAD.

OBSERVED AND MEAN VALUES TAKEN FROM FIG. 3.

Values on the two principal grades, and maximum, minimum, and mean values for entire trip of the Special Test-car, No. 4. August 17 and 18, 1891.

Grade on which reading was taken.	3.64% up.		3.94% up.		3.64% down.	Entire car route.		
Time and time-interval on grade.	11-37-30	11-31-45 to 11-33-30	12-8-0.	12-6-0 to 12-16-0.	12-30.	11-30 to 12-35.		
Conditions.	Steady.	Mean.	Steady.	Mean.	No Load.	Max.	Min.	Mean.
Coupled Turbines.								
Efficiency	49.1	47.4	48.5	46.7	22.3	49.5	−4.88†	34.4*
Total turbine friction H. P.	96.8	98.3	96.3	98.2	97.1	111.5	93.8	98.87
Turbine shaft revolutions per minute	97.0	98.5	97.0	101.5	103.0	122.0	96.0	103.5
Variations of power from hydraulic H. P. at station to electrical H. P. delivered to car.								
Hydraulic H. P.	190.0	187.0	188.2	184.2	124.9	195.0	116.3	150.50
Turbine H. P.	93.2	88.7	91.4	86.0	27.8	95.6	−5.67†	51.75
E. H. P. at station	48.5	45.9	47.6	44.7	0.0	63.9	0.0	....
E. H. P. at car	46.0	43.1	44.8	42.3	0.0	59.4	0.0	....

* Calculated from integrated means of horse power curves for entire run. All other values in this column are integrated means.

† No. 2 turbine kept at fixed gate opening of 17.4%. The negative value of the turbine H. P. arises from this turbine driving the No. 1 turbine, at the sudden change of load, indicated ; so that the No. 1 turbine output, the total turbine output, and the efficiency are all negative.

TABLE V.

PERFORMANCE OF TURBINES AND DYNAMOS UNDER LIGHT LOAD.

SUMMARY OF POWER DISTRIBUTION AND EFFICIENCIES.

With Special Test-car, No. 4, on the two principal grades, 3.64% and 3.94%, no other car being on the road at that time, August 17 and 18, 1891.

Grade on which reading was taken	*3.64%		**3.94%	
Time and time interval on grade	11-37-30	11-31-45 to 11-39-30	12-8-0.	12-6-0 to 12-16-0.
Conditions.	Steady.	Mean.	Steady.	Mean.
Speed of turbine shaft	97.0	98.5	97.0	101.5
Gate opening, No. 1 turbine	31.7	30.6	31.0	30.2
" No. 2 turbine (fixed gate)	17.4	17.4	17.4	17.4
Speed of car,† miles per hour	12.4	11.43	11.1	10.62
Power delivered :				
Hydraulic H. P. delivered to turbines	190.0	187.0	188.2	184.2
Turbine H. P. delivered to transmission machinery	93.2	88.7	91.4	86.0
H. P. delivered to dynamo	56.1	53.15	55.1	51.9
E. H. P. delivered to line	48.5	45.9	47.6	44.7
E. H. P. delivered to car motors‡	46.0	43.1	44.8	42.3
H. P. delivered at car axles	34.50	31.92	33.7	31.78
Power lost :				
Turbine friction H. P.	96.8	98.3	96.8	98.2
Transmission Machinery Friction, H. P	**37.1**	**35.55**	**36.3**	**34.1**
Dynamo losses	7.6	7.25	7.5	7.2
Line losses	2.5	2.8	2.8	2.4
Car motor, gearing and axle losses	11.5	11.18	11.1	10.52
Total losses, Hyd. H. P. to car axle H. P.	155.5	155.08	154.5	152.42
H. P. available at car axle	34.5	31.92	33.7	31.78
Total Hyd. H. P. delivered	190.0	187.0	188.2	184.2
Proportional parts of full load :				
Hyd. load, per cent. of full Hyd. H. P.	36.2	35.7	35 9	35.2
Turbine load, per cent. of full turbine H. P.	22.5	21.4	22.1	20.6
Dynamo load, per cent. of full dynamo output	36.2	34.2	35.5	33.3
Motor load, per cent. of full motor output	73.5	68.8	71.5	67.7
Percentage distribution :				
Turbine friction, per cent. loss	.509	.526	.514	.533
Transmission Machinery Friction, per cent. loss	**.195**	**.187**	**.194**	**.185**
Dynamo, per cent. loss	.040	.038	.040	.039
Line, " "	.013	.015	.014	.013
Car motor, gearing and axle losses	.061	.063	.059	.057
Available power at car axle	.182	.171	.179	.173
Total hydraulic power	1.000	1.000	1.000	1.000
Summary of efficiencies :				
Efficiency of No. 1 turbine	58.7	57.4	58.0	55.8
" No. 2 turbine	33.8	32.9	34.0	31.4
" coupled turbines	49.1	47.4	48.5	46.7
Efficiency of Transmission Machinery	**60.2**	**59.9**	**60.3**	**60.4**
" No. 1 dynamo ¶	86.5	86.3	86.4	86.1
" line, mean, for given grade	94.8	94.0	94.1	94.8
Efficiency of car motors, from terminals to car axle	75.0	74.2	75.2	75.1
Efficiency from dynamo terminals to car axle	71.2	69.7	70.8	71.2
Efficiency from Hyd. H. P. to line H. P.	25.5	24.5	25.3	24.2
Efficiency from Hyd. H. P. to car axle H. P.	18.2	17.1	17.9	17.2

* On 3.64% grade car started under normal and almost steady conditions. Record ends without stopping the car.

** On 3.94% grade car started from a full stop, stopped twice *en route*, and almost stopped at end of grade.

† Mean speeds on grade are exclusive of stops.

‡ Car equipped with two No. 12 Edison S. R. 25 H. P. motors, style B.

¶ 100 K. W. Edison railway generator, rated at 650 revolutions per minute.

Weight of car alone, 22,000 lbs.; weight of car and load during test, 22,500 lbs.

Plant driven by two 250 H. P. 45-inch Hercules turbines, coupled; one of which, No. 2, is kept at fixed gate opening of 17.4 per cent.

TABLE VI.

PERFORMANCE OF THE TURBINES AND DYNAMOS UNDER HEAVY LOAD.

OBSERVED AND MEAN VALUES TAKEN FROM FIG. 5.

Values on the two principal portions of the run ; and maximum, minimum and mean values for the entire run. Aug. 20, 1891.

Section in which reading is taken	I.	II.	Total.
Time base of section, hours and minutes	10.30—11.01	11.01—11.14	10.30—11.14.
Time interval of section, in minutes	31	13	44
Efficiency, maximum	58.0	**69.0**	69.0
" minimum	26.0	39.0	26.0
Mean (from mean H. P's)	46.6	59.4	**51.1**
Turbine friction H. P.			
Maximum	110.0	99.0	110.0
Minimum	82.0	86.0	82.0
Mean (integrated)	97.6	95.1	96.8
Speed of turbine shaft, revolutions per minute.			
Maximum	**113.0**	104.5	113.0
Minimum	90.5	**78.0**	78.0
Mean (integrated)	101.8	93.5	**99.4**
Hydraulic H. P., maximum	230.0	**300.0**	300.0
" " minimum	140.0	162.0	140.0
Mean (integrated)	182.7	234.0	**197.8**
Turbine H. P., maximum	132.0	**208.0**	208.0
" " minimum	37.0	64.0	37.0
Mean (integrated)	85.1	138.0	**101.0**
Electrical H. P., maximum	76.0	**124.0**	124.0
" " minimum	0.0	40.0	0.0
Mean (integrated)	38.5	82.0	**51.3**

TABLE VII.

PERFORMANCE OF TURBINES AND DYNAMOS UNDER HEAVY LOAD.

SUMMARY OF POWER DISTRIBUTION AND EFFICIENCIES.

Section in which reading is taken	I.	II.	Total.
Time base of section, hours and minutes	10.30—11.01	11.01 11.14	10.30—11.14
Time interval of section, in minutes	31	13	44
Speed of turbine shaft, revolutions per minute:			
Maximum	**113.0**	104.5	113.0
Minimum	90.5	**78.0**	78.0
Mean (integrated)	101.8	93.5	**99.4**
Gate opening, proportional part of the full opening of the speed-gate, per cent.			
No. 1 Turbine, maximum	38.0	**41.0**	41.0
minimum	**6.5**	8.5	6.5
Mean (integrated)	23.4	25.5	**24.0**
No. 2 Turbine, maximum	44.0	**66.0**	66.0
minimum	**9.5**	16.5	9.5
Mean (integrated)	25.0	38.7	**29.1**
Power Delivered:			
Hydraulic H. P. delivered to turbines	182.7	234.0	197.8
Turbine H. P. delivered to transmission machinery	85.1	133.9	101.0
H. P. delivered to dynamos	50.5	95.8	63.7
E. H. P. delivered to line	38.5	82.0	51.3
E. H. P. " " car motors	36.2	77.1	43.2
H. P. " " car axles	25.4	54.0	33.7
Power Lost:			
Turbine friction H. P	97.6	95.1	96.8
Transmission Machinery Friction, H. P	**34.6**	**43.1**	**37.3**
Dynamo losses	12.0	13.8	12.4
Line losses	2.3	4.9	3.1
Car motor and gearing losses	10.8	23.1	14.5
Total losses, Hyd. H. P. to car axle H. P	157.3	180.0	164.1
H. P. available at car axle	25.4	54.0	33.7
Total hydraulic H. P. delivered	182.7	234.0	197.8
Proportional Parts of Full Load:			
Hydraulic load, per cent. of full hydraulic horse-power	35.1	45.0	38.0
Turbine load, per cent. of full turbine horse-power	20.4	33.4	24.3
Dynamo load, per cent. of full dynamo output	14.4	30.6	19.2
Motor load, per cent. of full motor output	14.5	24.6	17.2
Percentage Distribution:			
Turbine friction, per cent. loss	0.534	0.406	0.489
Transmission Machinery Friction, per cent. loss	**0.189**	**0.184**	**0.189**
Dynamos, in parallel, per cent. loss	0.066	0.059	0.063
Line losses, mean of total length	0.013	0.021	0.016
Car motor, gearing and axle losses	0.059	0.099	0.073
Available power at car axle	0.139	0.231	0.170
Total hydraulic power	1.000	1.000	1.000
Summary of Efficiencies:			
Efficiency of coupled turbines	46.6	59.4	51.1
Efficiency of Transmission Machinery	**59.3**	**69.0**	**63.1**
" dynamos in parallel	76.3	85.2	80.5
" line, mean of total length	94.0	94.0	94.0
" car motors	70.0	70.0	70.0
" from Hyd. H. P. to line H. P	21.1	35.0	26.0
" from Hyd. H. P. to car axle H. P	13.9	23.1	17.1

Efficiencies from Hydraulic H. P.	To line H. P.	To car axle H. P.
Efficiency at half load of dynamos, which is 50% turbine capacity, or 57% full Hyd. H. P.	45.0	29.8
Efficiency at full load of dynamos, which is 95% turbine capacity, or 93% full Hyd. H. P	55.2	36.5

Cars equipped with two No. 12 Edison S. R., 25-H. P. motors, style B. Current supplied by two 100-K. W. Edison railway generators, rated at 650 revolutions per minute.

Plant driven by two 250-H. P., 45-inch Hercules turbines, coupled. Four 22,000-lb. cars out on road, in service during test.

(4) POWER LOSSES IN A CABLE PLANT.

This test was made on the Druid Hill Avenue Cable Plant, Baltimore, Md., jointly by Mr. Hermann S. Hering and the author, March 29, 1892, while associated in the department of electrical engineering at the Johns Hopkins University.

A new plan is employed to drive the cable drums by not having them geared together; but, using in connection with each driving drum, a Whitton compensator. This consists of a simple box-train epicyclic gear. The idler drum is double, one-half of which is mounted loose on the shaft, while the other half is keyed to the same.

The cable drums are placed, one on each side of the rope-drive drum, on the main shaft, and driven through positive clutch couplings. These as well as similar clutches on the engine shaft, are operated by hand, through compound gear.

In the eleven miles of single track, there are fifty-eight single track curves, making 2,900 feet of curve construction in all..

The power house, at the other end of the line, is a duplicate of the Druid Hill Avenue one.

The above particulars are taken from the *Street Railway Journal*, March, 1891, which contains a full and illustrated description of the "Baltimore Cable Road" and the Druid Hill Avenue power house.

The test was made under the ordinary working conditions of the road, the usual daily traffic and operations not being interfered with in the least during the normal running time of 24 hours. The plant consisted of two Wetherill-Corliss type of engines, rated at 414 H.P., at 60 revolutions, 80 lbs. boiler pressure, and $\frac{1}{5}$ cut-off. Rope-driving is used to transmit the power from the engine shaft to the cable-driving shaft. Only one engine was run during the test. It was fitted with pantograph reducing motion, and two Thompson indicators. The method of indicating the engine was to take simultaneous cards from each end of the cylinder every 2 minutes. The pencil was kept on for a single revolution only, and the same card was used for five consecutive observations, at 2-minute intervals each. New cards were put on every ten minutes, so that each card contained a record of five single diagrams from one end of the cylinder. Sometimes the pencil, at one end of the cylinder, would be left on the card for 3 or 5 minutes, while the single cards were taken from the other end as usual. On comparing the two cards it was found that the mean obtained from the five single cards was practically the same as that from the one continuous card. The method therefore followed, in taking the cards and averaging them, gave a satisfactory value of the mean indicated horse-power of the engine during the interval. The sum of the areas of the five single diagrams was divided by 5, giving the mean area, which divided by the mean length gave the mean effective pressure for the 10-minute interval.

The frictional losses of the engine, drum, and cables were determined by running the engine with only that part of machinery attached, of which it was desired to obtain the friction horse-power. It will be noticed that the sum of the individual friction horse-powers of the large and small cables is not equal to that required to drive the two together. This shows an increase in the friction of the transmission machinery and through the rope drive, a result which is to be expected when the transmitted load is increased and the two cable-driving shafts are operated as one.

The power necessary to drive the cable, under no load, and at various speeds, was obtained by a special test; from the plotted results of this run it appears that the cable friction horse-power varies almost directly with the speed at which the cable is running.

It is the extent of curved track and the steep grades on the road, which no doubt make the efficiency of the cables so low; for, as will be seen by referring to the summary of efficiencies, only 45.9% of the power delivered to both cables is available for use in propelling the cars.

The data and particulars involving the power losses in the transmission machinery are tabulated below:

TABLE NO. VIII.

DATA AND PERFORMANCE OF ENGINE AND CABLE.

Dimensions of Engine.

Diameter of cylinder	28 inches.
Stroke of piston	5 feet.
Piston stroke to release (proportional part of stroke)	100.0 stroke.
Piston stroke to exhaust closure	head, 92%; crank, 92%
Clearance from card	head, 2.7%; crank, 3.0%

Dimensions and Data of Transmission and Cable Machinery.

Driving drum, on engine shaft	8 feet diameter, 8 feet 4-inch face.
Main shaft drum	24 “ “ “ “ “
Number of ropes in rope-drive	32
Main shaft, between clutch couplings	18 inches diameter.
Main shaft, between clutch couplings and the cable-driving drums	15 “ “
Cable drums, down-town	10 feet 6 inches diameter
“ “ up-town	16 “ diameter.
Cable speeds, down-town	7½ miles per hour.
“ “ up-town	11½ “ “

Data from Test.

Date of test	March 29, 1892.
Duration of test, normal running time of engine	21 hours, 52 minutes.
Mean boiler pressure, pounds per gauge	80.96 lbs.
Ratio of boiler to initial engine pressure	88.3%
Mean speed of engine	60.53 revolutions per minute.
Mean piston speed	605.3 feet per minute.
Average mean effective pressure	head end, 20.6 lbs. crank “ 22.4 “
Mean total indicated horse-power	241.7 H.P.
Mean back pressure in ports	head end, 1.91 lbs. crank “ 1.43 “
Mean back pressure of compression	head end, 21.0 lbs. crank “ 22.0 “

TABLE IX.

SUMMARY OF POWER DISTRIBUTION AND EFFICIENCY OF CABLE PLANT.

Power Developed and Delivered.

Horse-power developed, cars on, maximum 6.30 P.M.	342.2	H.P.
“ “ “ minimum 5.04 A.M.	156.0	“
“ “ “ mean (of run)	241.7	“
Horse-power developed without cars on, but with both cables, drum, and engine friction, maximum at start	174.6	“
Horse-power developed without cars on, but with both cables, drum, and engine friction, minimum	148.8	“
Horse-power developed without cars on, but with both cables, drum, and engine friction, mean	152.35	“
Horse-power developed without cars on, but with down-town cable, drum, and engine friction, mean	124.2	“
Same, but with up-town cable, drum, and engine, mean	69.0	“
Same, but with drum and engine only, mean	47.45	“
Same, but with engine only (engine uncoupled), mean	29.15	“
Horse-power delivered by engine shaft	212.55	“
Horse-power delivered by main-driving shaft from clutch couplings	194.25	“

Horse-power delivered by both cables to cable car grips	89.35 H.P.
Friction horse-power of both cables and additional shafting required by same, mean	104.9 “
Friction horse-power of both cables, alone, mean	98.3 “

Power Lost.

Engine friction horse-power	29.15 H.P.
Drum friction	18.30 “
Up-town cable friction	21.55 “
Down-town cable friction	76.75 “
Shafting, due to mean load on both cables	6.60 “
Total friction horse-power	152.35 H.P.
Mean available horse-power delivered to cars	89.35 “
Mean total indicated horse-power developed by engine	241.70 H.P.

Summary of Percentage Distribution of Power.

Percentage lost in engine friction	12.10%
“ “ drum friction	7.60%
“ “ up-town cable friction	8.90%
“ “ down-town cable friction	31.80%
“ “ shafting (due to use of both cables)	2.70%
Percentage available power at cars	36.90%
Total power developed by engine	100.0%

Summary of Efficiencies.

Mechanical efficiency of engine	87.9%.
Efficiency of rope-drive transmission, from engine shaft to main-shaft clutch couplings	**91.4%.**
Efficiency of both cables and cable-driving drums, from main-shaft clutch coupling	46.0%.
Efficiency of the cable plant, from the indicated horse-power of engine to power available at cars	36.9%.

(5) POWER LOSSES IN TRANSMITTING THROUGH A COUNTERSHAFT.

These tests were made to determine the losses in the usual method of driving dynamos; namely, through two belted connections and a countershaft. They were made jointly by Mr. Clement R. Jones and the author, in the Mechanical Engineering Department of the West Virginia University, in March, 1894.

Two sets of tests were made:

(1) To determine the engine friction.

(2) To determine the transmission machinery friction.

(1)—*In the engine friction tests,* 30-minute runs were made, and indicator cards and speed of engine taken every minute. The value of the engine friction under load is the mean of four 30-minute runs, with water-cooled Prony brake and platform scales for measuring the power, with minute observations, as in the first case. The results are given below.

TABLE X.

DATA AND PERFORMANCE OF ENGINE.

Dimensions of Engine.

Engine, Wheelock, rated horse-power, at 75 revolutions, 80 lbs. initial pressure, gauge, ¼ cut-off	40 H. P.
Diameter of cylinder	10¾ inches.
Stroke of piston	30 inches.
Crank shaft, crank bearing	4½ inches diameter, 11 inches long.
" " outboard bearing	4½ inches diameter, 10 inches long.
Band-wheel; diameter on crown	96⅝ inches; face, 17 inches.

Engine Performance.

Engine running free,—column (*a*).
Engine running under Prony brake load,—column (*b*).

	(*a*)	(*b*)
Time interval of run, minutes	30	30
Corrected mean boiler pressure	76.00	75.70
Mean revolutions, per minute	74.93	71.18
Mean indicated horse-power	2.77	17.18
Mean Prony brake horse-power	0.00	14.57
Engine friction, horse-power	2.77	2.61
" " per cent	100.00	15.19
Mechanical efficiency of the engine, per cent.	0.0	84.81

(2)—*In the transmission machinery friction tests,* two separate runs were made, one without and the other with leather covering on dynamometer pulley, as noted in the following table under (*c*) and (*d*) respectively. The power was transmitted through a countershaft and absorbed by an Alden dynamometer, as shown in Fig. 211, on which the principal dimensions are also given.

It was possible to maintain a very steady load on the engine by means of the Alden dynamometer, and its constant was carefully determined and allowed for in the calculations. It is a 32-inch 50-H. P. dynamometer, especially designed and con-

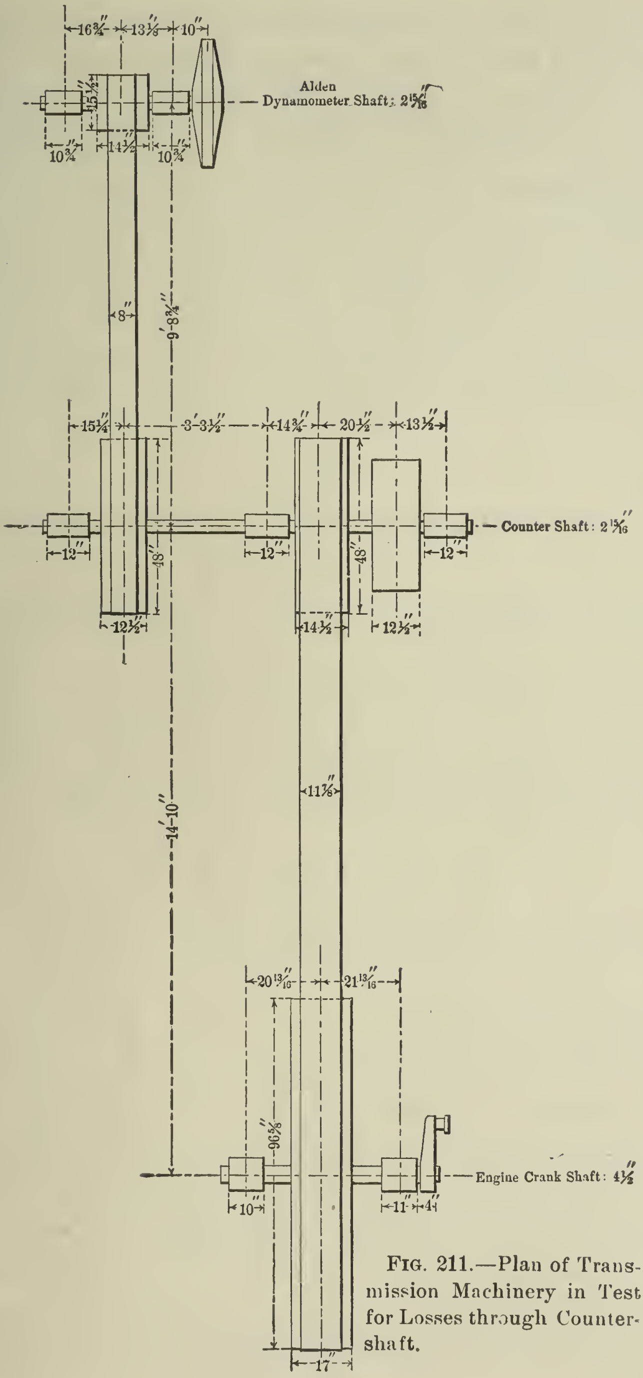


FIG. 211.—Plan of Transmission Machinery in Test for Losses through Countershaft.

structed for use with high-water pressure at high speeds; but, during this test, it was worked far below its limit, as small powers only were absorbed. Its position in the test is that which a dynamo would occupy in the ordinary electric light plant, driven by a slow-speed engine. It is believed that tests of this character will serve to show the ordinary losses in such installations.

Indicator cards and speeds of engine and dynamometer shafts were taken every two minutes. The engine friction horse powers, tabulated for (*c*) and (*d*), were taken from a curve plotted from the previously recorded results, on the usual assumption that the engine friction varies directly as the speed, regardless of the load.

TABLE XI.

DATA AND PERFORMANCE OF THE TRANSMISSION MACHINERY.

Dimensions and data.

Band-wheel of engine—diameter on crown	96⅝ inches; face, 17 inches.
Countershaft pulleys, driven—diameter on crown	48 " " 14½ inches.
" " driver " "	48 " " 12½ inches.
Dynamometer pulley (*c*) bare cast-iron: diameter on crown	15¼ inches; face, 14½ inches.
Dynamometer pulley (*d*) covered with Shultz patent leather covering: diameter on crown	15½ inches; face 14½ inches.
Main driving belt, light double, oak-tanned	11⅞ × ⅓ inch.
Dynamometer belt, single, oak-tanned	8 × 7/32 inch.

Transmission Machinery Performance.

Plain cast-iron pulley on dynamometer,—column (*c*).
Shultz patent leather covering on dynamometer pulley,—column (*d*).

	(*c*)	(*d*)
Time interval of run, minutes	60	30
Corrected mean boiler pressure	59.29	64.5
Mean revolutions per minute, engine	67.0	68.25
" " " " dynamometer	402.60	417.06
Calculated speed of dynamometer, ratio	6.34	6 234
" " " " speed	424.8	425.5
Loss in speed, from engine to dynamometer shaft	22.2	8.44
Slip of belt, per cent. of calculated speed	5.23	1.98
Tension at which belts were laced and run, average of initial, and final tensions on belts—pounds per square inch:		
Main driving belts	40	80
Dynamometer belt	75	135
Mean indicated horse-power	19.122	13.21
Engine friction, horse-power (from plotted curve)	2.50	2.54

Horse-power delivered to transmission machinery	16.622	10.670
Dynamometric horse-power absorbed	15.275	9.960
Horse-power lost in transmission machinery	**1.347**	**0.710**
Frictional loss, per cent., engine	13.08	19.23
Frictional loss, per cent., transmission machinery	**8.10**	**6.65**
Efficiency, per cent., engine	86.92	80.77
Efficiency per cent., transmission machinery	**91.90**	**93.35**

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